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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : F02B 29/04	A1	(11) International Publication Number: WO 98/55744 (43) International Publication Date: 10 December 1998 (10.12.98)

(21) International Application Number: PCT/KR98/00141

(22) International Filing Date: 3 June 1998 (03.06.98)

(30) Priority Data:
1997/23511 7 June 1997 (07.06.97) KR

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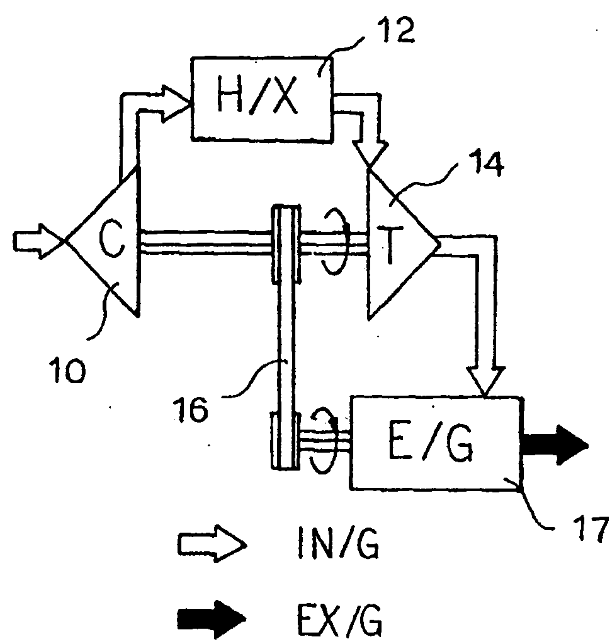
(81) Designated States: CN, JP, US, European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE).

Published
With international search report.
Before the expiration of the time limit for amending the claims and to be republished in the event of the receipt of amendments.

(54) Title: CHARGER FOR AN INTERNAL–COMBUSTION ENGINE

(57) Abstract

A charger for an internal–combustion engine is disclosed. The charger includes a compressor (10) for compressing supplied air to a high temperature/high pressure state and emitting the same; a heat exchanger (12) for reducing a temperature of the high temperature/high pressure air received from the compressor to convert the same into a low temperature/high pressure state; and a turbine (14) for expanding the air received in a low temperature/high pressure state from the heat exchanger to perform work externally such that the temperature of the air is further reduced, after which the air is supplied to the internal–combustion engine (17).



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CHARGER FOR AN INTERNAL-COMBUSTION ENGINE

Technical Field

The present invention relates to a charger for an internal-combustion engine, and more particularly, to a charger for an internal-combustion engine which increases an amount of intake air to improve power density and efficiency of the internal-combustion engine.

Background Art

There are many different types of internal-combustion engines including the gasoline engine, diesel engine, rotary engine and jet engine. Among these, the gasoline engine is most commonly used in automobiles to generate driving force for the same. In the gasoline engine, a mixture of air and fuel is supplied to one or more cylinders to begin a continuously repeated and sequential process of intake, compression, explosion and exhaust to generate drive power. Diesel engines, also commonly used in automobiles, utilize a process of air intake, air compression, fuel injection, explosion and exhaust.

The volume of the air and fuel mixture supplied to the cylinder must be maintained at a maximum level to optimize

engine power. The charger is able to facilitate this function.

Chargers maximize the amount of air supplied to the engine and the amount of fuel mixed with the air such that engine power is greatly enhanced. There are two types of chargers: turbo chargers and super chargers. Turbo chargers utilize a turbine which rotates by force created by the exhaust of gases to rotate a compressor such that the amount of air supplied is increased, while in super chargers, a compressor is directly mounted on an output shaft of the engine to compress air then supply the same to the cylinder.

Such chargers are effective in improving engine efficiency by increasing power density, and in the case of diesel engines, are utilized to realize the additional advantages of increasing output and reducing exhaust gas pollutants. However, chargers employed in present-day internal-combustion engines experience temperature increases as a result of rising levels of pressure caused by the process of compressing air.

Accordingly, a temperature of air exiting the charger increases such that engine knocking occurs, thereby decreasing a compression ratio of the engine. Further, this increase in the temperature of air exiting the charger results in an excessive amount of fuel being fed to the cylinders, which, in turn, causes a delay in ignition timing. Both these

drawbacks negatively influence overall engine efficiency, and with regard to decreases in the compression ratio, charge pressure is reduced resulting in an insufficient generation of power (p. 870-871, Internal Combustion Engine Fundamentals, John B. Heywood).

In addition, prior chargers increase a temperature of a combustion chamber to a level higher than that required such that a considerable amount of nitrous oxide (NO_x) is generated by the resulting high-temperature combustion.

To remedy these problems, prior chargers utilize an inter-cooler to reduce the temperature of the air compressed by the charger. However, because such inter-coolers, of which there are water-cooled and air-cooled types, are unable to reduce the temperature of the air to a level below 60°C, improvements made in engine power density and efficiency are minimal.

Prior art related to the present invention include U.S. Patent No. 5119795, Intake System with Mechanical Supercharger for Internal Combustion Engine; U.S. Patent No. 4993228, Internal Combustion Engine with Turbosupercharger; U.S. Patent No. 5269143, Diesel Engine Turbo-expander; U.S. Patent No. 4774812, Turbocharged Engine; and U.S. Patent No. 4674283, Turbocharging System for an Internal Combustion Engine.

Among the above patents, U.S. Patent No. 4993228 relates

to a closed cooling system utilizing coolant and including a compressor and a turbine to lower intake temperatures of a charged diesel engine to reduce nitrous oxide (NO_x) emissions, while the remainder of the above patents relate to control methods for conventional turbochargers and superchargers. However, U.S. Patent No. 4993228 is directed for use in only diesel engines, and its structure is such that manufacture of the same is complicated and difficult.

Summary of the Invention

The present invention has been made in an effort to solve the above problems.

It is an object of the present invention to provide a charger for an internal-combustion engine which improves power density and efficiency of internal-combustion engines, and reduces the emission of nitrous oxide and other such pollutants.

To achieve the above object, the present invention provides a charger for an internal-combustion engine. The charger includes a compressor for compressing supplied air to a high temperature/high pressure state and emitting the same; a heat exchanger for reducing a temperature of the high temperature/high pressure air received from the compressor to convert the same into a low temperature/high pressure state;

and a turbine for expanding the air received in a low temperature/high pressure state from the heat exchanger to perform work externally such that the temperature of the air is further reduced, after which the air is supplied to the internal-combustion engine.

According to a feature of the present invention, the compressor and the turbine are mechanically connected, and a power transmitter is further provided, the power transmitter receiving drive power needed to drive the compressor.

In another aspect, a nozzle area of the turbine is variable, and the turbine is able to perform a function of a throttle valve.

In yet another aspect, the compressor and the turbine are mechanically connected, and an additional compressor is provided.

Brief Description of Drawings

The present invention will become more fully understood from the detailed description given hereinbelow and the accompanying drawings which are given by way of illustration only, and thus are not limitative of the present invention, and wherein:

FIG. 1 is a schematic diagram of a charger for an internal-combustion engine according to a first preferred

embodiment of the present invention;

FIG. 2 is a schematic diagram of a charger for an internal-combustion engine according to a second preferred embodiment of the present invention;

FIG. 3 is a graph illustrating a relation between charge temperature and charge pressure;

FIG. 4 is a graph illustrating a relation between compression ratio and charge pressure;

FIG. 5 is a graph illustrating a relation between spark advance and charge pressure;

FIGS. 6a to 6d illustrate differing states of a variable area nozzle used in a turbine of a charger according to the present invention;

FIG. 7 is a schematic diagram of a charger for an internal-combustion engine according to a third preferred embodiment of the present invention; and

FIG. 8 is a schematic diagram of a charger for an internal-combustion engine according to a fourth preferred embodiment of the present invention.

Best Mode for carrying Out the Invention

Preferred embodiments of the present invention will now be described in detail with reference to the accompanying drawings.

Referring first to FIG. 1, shown is a schematic diagram of a charger for an internal-combustion engine according to a first preferred embodiment of the present invention. In this drawing and all subsequent drawings illustrating different inventive embodiments, IN/G indicates intake of air into an engine, while EX/G indicates exhaust of air from the same.

As shown in FIG. 1, the charger according to the first embodiment comprises a compressor 10 which compresses air received from outside the charger to a high temperature/high pressure state; a heat exchanger 12 for reducing a temperature of the air compressed to a high temperature/high pressure state by the compressor 10 such that the air is converted into a low temperature/high pressure state; a turbine 14 for expanding the air converted into a low temperature/high pressure state by the heat exchanger 12, thereby further lowering the temperature of the air, and supplying the air to an engine 17; and a power transmitter 16 for transmitting drive power of the engine 17 to the compressor 10.

The heat exchanger 12 is either an air or water-cooled type, and the turbine 14, expanding the compressed air to lower the temperature of the same, generates drive power by this process of expanding the air to drive the compressor 10 or to be supplied to the engine 17, thereby reducing energy loss.

The operation of the turbine 14 will be described in more detail hereinafter.

The first law of thermodynamics can be expressed as follows:

Q (external heat transmission) = ΔU (internal energy variations) + W (work performed externally by a gaseous body)

However, as engine intake air moves at a rate of roughly 30-40m/sec, external heat transmission is negligible. Thus, Q can be said to equal 0, and if it is assumed that air is the above gaseous body, the equation, $\Delta U = C_v \times \Delta T$, results.

Accordingly, it can be known that $\Delta T = -W/C_v$, and when work is transmitted externally during a process of isolating heat, the temperature of gaseous material is reduced. That is, if high pressure air passing through a heat exchanger is supplied to a turbine to be expanded and perform work externally, a corresponding amount of internal energy is reduced such that the temperature of the gaseous body is lowered.

The operation of the charger according to the first embodiment and structured as in the above will now be described.

The compressor 10, driven by power received from the engine 17 via the power transmitter 16, intakes external air to compress then supply the same to the heat exchanger 12.

Here, as the temperature of the air increases from the compression of the same, the air is discharged from the compressor in a high temperature/high pressure state. The heat exchanger 12 lowers the temperature of the air received from the compressor 10 to convert the air into a low temperature/high pressure state, after which the heat exchanger 12 supplies the air to the turbine 14.

The turbine 14 expands the air received in a low temperature/high pressure state from the heat exchanger 12 then supplies the air to the engine 17. Here, the high pressure air performs work externally while being expanded such that the temperature of the air is reduced by as much of a degree as the work performed. As a result, the intake temperature of the engine is further reduced.

Accordingly, because the amount of air able to be supplied to the engine 17 is increased and the temperature of the air is at or below a suitable level, the engine is able to operate under conditions of a high compression ratio, a maximum air fuel ratio and an optimum ignition timing, without the occurrence of engine knocking. As a result, engine efficiency is enhanced and output of the same is increased.

In the first embodiment, the compressor 10 receives drive power from the engine 17 via the power transmitter 16. Here, as the compressor 10 receives this drive power (W_1) from the

engine 17, expansion force (W_2) generated as the turbine 14 expands the compressed air operates such that actual engine power loss is W_1 - W_2. However, as engine output is improved by the increase in intake air, overall output is raised.

Referring now to FIG. 2, shown is a schematic diagram of a charger for an internal-combustion engine according to a second preferred embodiment of the present invention. As shown in the drawing, a compressor 20 is driven by a motor 26, and air compressed by the compressor 20 is cooled by a heat exchanger 22 then supplied to a turbine 24. The turbine 24 expands the received air to further lower a temperature of the same, after which the air is supplied to an engine 27.

As described in the first embodiment, the inventive charger improves the output of the internal-combustion engine. Specific related data are shown in Chart 1 below.

[Chart 1] An example of performance for a theoretical charger (first embodiment).

Measured item	Results	Unit
Charger intake temp.	288.15	°K
Charger intake press.	101325 (1 atmos. press.)	Pa
Charger intake density	1.225	Kg/m^3
Comp. output pressure	303975 (3 atmos. press.)	Pa
Charger output temp.	420	°K
Charger output temp.	273	°K
Charger output press.	155946 (1.5)	Pa

Measured item	Results	Unit
Charger output density	1.99	Kg/m^3
Comp. work consumption	25	kW
Turbine work generation	9.3	kW
Charger pure work util.	15.7	kW
Engine output	217	kW
Engine pure output	201	kW

(In the case of a 2000CC, 109Kw engine, compressor efficiency is 0.8, turbine efficiency is 0.9 and heat exchanger efficiency is 0.75.)

As shown in Chart 1 above, an 84% increase in output, (201-109)/109 can be expected at a 0.5 atmospheric pressure charge (1.5 atmospheric pressure). In particular, intake temperature becomes 0°C (273°K). Using the conventional charging method (the prior turbocharger method), in which an inter-cooler is utilized, charger pressure is raised roughly 0.4 atmospheric pressure to realize an increase in output of

approximately 40%.

For additional related information, the Miller Cycle, used to improve the conventional charge method, will be described hereinafter.

The Miller Cycle is a method in which an intake valve is closed during intake to realize partial expansion and control the compression ratio during the compression process. Basically, the method is used to realize a variable compression ratio engine, and as the Miller Cycle aims to increase cycle efficiency by only reducing the compression ratio of the engine, the compression ratio is actually reduced as a result of the closing of the intake valve during the intake cycle. On the other hand, as the expansion ratio is unchanged, output from 'pure work = expansion work - compression work' is increased. However, by the closing of the intake valve during the intake cycle, a sufficient amount of air is not able to be supplied such that volummetric efficiency is reduced. This can be offset by increasing charge pressure.

For example, in the case of reducing the compression ratio by half to prevent engine knocking, when using ordinary fuel, it is not possible to use a level of charge pressure higher than absolute pressure 1.8 atmospheric pressure as a result of engine knocking limitations. Accordingly, the amount of intake air is, in fact, reduced such that effects of charging are eliminated. As a result, to use the Miller Cycle in a gasoline engine, various factors are unavoidable

including ignition timing retard and an increase in the air-fuel ratio, and the required use of premium gasoline (Mazda vehicles utilizing the Miller Cycle method require that premium gasoline be used).

Therefore, although in principle the Miller Cycle attempts to reduce the compression ratio to delay knocking by interrupting the intake cycle, this operation is not possible in maximal conditions. Further, as volummetric efficiency is reduced by this interrupting of the intake cycle, the effects of charging are reduced (e.g. Mazda's Millennium S realizes only a 24% increase in output compared to vehicles having similar engine sizes).

That is, in the Miller Cycle, if the intake valve is shut off before the intake cycle is finished, the expansion ratio is left unchanged while the compression ratio is reduced to prevent knocking during the compression cycle and reduce the work consumption during the compression cycle such that output is increased.

As a reference, the occurrence of knocking in a gasoline engine when turbocharging will be further described with reference to Figs. 3, 4 and 5 (derived from Internal Combustion Engine Fundamentals, page 871, John B. Heywood, McGraw-Hill). In the drawings, RON indicates an octane number of gasoline, ϕ indicates an air-fuel ratio, r_c indicates a compression ratio, With CAC is with charge air cooling, Without CAC is without charge air cooling, bmep is brake mean effective pressure and p_i indicates intake pressure.

As can be seen in FIG. 3, which is a graph illustrating a relation between charge temperature and charge pressure, when the compression ratio of an engine is 7, there is a limit to charge pressure as a result of the gasoline octane number, the amount of fuel injected and the temperature of supplied air, and the intake temperature must be lowered for the charge pressure to be increased.

Namely, to realize charging, the compression ratio of the engine must be lowered to around 7 to lower the maximal temperature during the compression cycle such that engine knocking does not occur. However, the lowering of the compression ratio sharply reduces engine efficiency. In an ideal engine, engine efficiency is 0.602 when the compression ratio is 10, while engine efficiency is 0.54 when the compression ratio is 7. That is, if an engine is charged while the compression ratio is maintained at a high level, an 11% increase is realized over conventional charged engines.

Further, as is shown in the graph of FIG. 3, if the gasoline octane number RON is high, charge pressure is increased, and if the air-fuel ratio is increased higher than a maximal air-fuel ratio (from $\phi = 1$ to $\phi = 1.1$), the charge pressure is increased (the effects of cooling using fuel), but at a lower air-fuel ratio ($\phi = 0.9$), knocking occurs at even a low charge pressure. In addition, if the temperature of the air entering the engine after passing through the charger is high, the charge pressure can not be increased.

Referring now to FIG. 4, shown is a graph illustrating

a relation between compression ratio and charge pressure in engines with and without charge air cooling. Here, in the engine with CAC, the temperature of the air before entering the engine is cooled. As the data shown in the graph of FIG. 4 are the results of a test of what is actually possible, the compression ratio is low, the fuel volume is higher than a maximal value and the octane number of the fuel is high.

As shown in FIG. 4, the charge pressure is higher in engines utilizing CAC than those without CAC. Further if the compression ratio reaches a level of roughly 9, charging is not possible, regardless of all other factors (i.e. whether a CAC is provided, a maximal octane number of fuel is used, fuel is additionally injected, etc.). Hence, turbocharged gasoline engines do not use a high compression ratio.

Referring now to FIG. 5, illustrating a relation between spark advance and charge pressure, when charge pressure is high, a small amount of spark advance results such that a sufficient amount of time for the transmission of fuel energy is not given. Accordingly, work loss occurs. However, if an inter-cooler is used, efficiency is improved as more time is provided.

Therefore, with the use of the inventive internal-combustion engine charger, the compression ratio of the engine is raised such that engine efficiency, higher than that of conventional engines, is realized (a high compression ratio is not able to be used in conventional engines because of limitations resulting from knocking, and the efficiency of

gasoline engines is less than that of diesel engines as a result of a low compression ratio).

Further, the level of energy consumption is low during the compression process if the temperature of the intake air is low such that an improvement in output is realized, and as an excessive amount of pressure develops at the end of the compression cycle if charge pressure is high, mechanical limitations are encountered. However, in the present invention, as part of the charge pressure is changed by a reduction in temperature, the density of air is sufficiently raised at even low charge pressures, thereby realizing increased output.

In the above, although the first and second embodiments are described with an assumption that a typical butterfly valve-type throttle valve for controlling the mixture and the amount of air supplied to the engine is used, a variable area nozzle is also able to be used in the inventive turbine for performing throttle valve functions.

In more detail, as the throttle valve generates pressure loss in air passing therethrough to control the amount of intake air, it is also possible to substitute this function by varying a nozzle area of the turbine.

Referring to FIG. 6, shown are differing states of a variable area nozzle used in a turbine of the inventive charger. If such a turbine having a variable area nozzle is used, during partial throttle (as opposed to full throttle), a size of the turbine variable nozzle is reduced to accelerate

the air, and the air absorbs energy on rotating blades of the turbine. Accordingly, pressure energy is restored to axial energy to reduce pressure and the amount of flow, and, simultaneously, the temperature of the air is lowered.

As a result, pressure loss in the conventional throttle valve, operating to generate loss during partial throttle (a cause of inefficiency in gasoline engines), is utilized as useful energy to reduce intake temperature. That is, as the lowering of temperature prevents the occurrence of knocking even if the compression ratio of the engine is increased, the application of a high compression ratio is possible, and the temperature of the combustion process is lowered such that loss caused by the transmission of heat is reduced and engine efficiency is increased.

This is shown in Chart 2 below.

[Chart 2] Pressures and temperatures at each point during partial throttle when using a variable nozzle in the first embodiment appearing in FIG. 1.

	Intake	Post compressor	Post heat exchanger	Post turbine
Pressure (Pa)	101325.00	203591.24	193411.67	50662.50 (0.5atmos. press.)
Temp. (°C)	26.5	114.8	53.0	-35.3

As shown in Chart 2, even when charge pressure is low during partial throttle, when the throttle valve is removed and the area of the turbine variable nozzle reduced to lower the outlet pressure to 0.5 atmospheric pressure, the

temperature of the air supplied to the engine is roughly -35°C.

This is indicative of the ability to substitute the throttle valve with a turbine utilizing a variable area nozzle, and that this is, in fact, more effective.

Referring now to FIG. 7, shown is a schematic diagram of a charger for an internal-combustion engine according to a third preferred embodiment of the present invention. In the third embodiment, a variable area nozzle is used.

In the charger according to the third embodiment, air supplied to an engine 77 is compressed by a first compressor 70, the first compressor 70 receiving drive power for rotation from a first turbine 75 which uses exhaust gases of the engine 77 to generate power. The air compressed by the first compressor 70 is supplied to a first heat exchanger 72 which reduces a temperature of the received air, after which the air is supplied to a second compressor 71.

Next, the air is again compressed by the second compressor 71 then supplied to a second heat exchanger 73. The second heat exchanger 73 again reduces the temperature of the air then supplies the same to a second turbine 74. Here, the air is accelerated by a variable area nozzle provided in the second turbine 74, and performs work on a rotor of the turbine 74 such that the temperature of the air is again reduced, after which the air is supplied to the engine 77.

The variable area nozzle performs the function of a throttle valve (i.e., controlling the volume of air), and

increases the efficiency of the turbine such that pressure loss during partial throttle is converted to a reduction in temperature, thereby increasing engine efficiency.

Further, drive power generated in the second turbine 74, which receives compressed air and expands the same, is used to rotate the second compressor 71.

Using the charger of the third embodiment, as the driving of the compressor and turbine is more easily matched, the charger is able to operate more efficiently, a size of the compressor can be minimized, and lengths of connectors between the structural elements can be reduced. Chart 3 below illustrates various data related to engine efficiency with the use of the third embodiment.

[Chart 3] Comparison data of engine efficiency

		3rd Embodiment	Turbo-charger	Natural intake
Full Throttle	*EHE	0.45	0.40	0.42
	*VE	0.38	0.34	0.29
Partial Throttle	*EHE	0.38	0.34	0.35
	*VE	0.27	0.22	0.245

* EHE: Engine heat efficiency

* VE: Vehicle efficiency

As shown in FIG. 8, illustrating a schematic diagram of a charger for an internal-combustion engine according to a fourth preferred embodiment of the present invention, a first compressor is driven by axial power of an engine. Although efficiency is reduced with such a configuration as exhaust gases are not utilized, responsiveness is greatly improved.

The charger for an internal-combustion engine structured and operating as in the above has many advantages. Namely, as the engine operates without knocking in a state where a high compression ratio, a maximal ignition timing and a maximal air-fuel ratio are maintained, a 20% improvement in engine efficiency and a 100% improvement in output are simultaneously realized; it is possible to apply the inventive charger to conventional gasoline engines; the inventive charger is substantially lighter in weight compared to conventional chargers; the inventive charger increases the efficiency of the internal-combustion engine as it enables a high output-to-weight ratio; the inventive charger reduces the internal temperature of the engine such that the life span of the same is increased; and the inventive charger reduces the temperature of combustion to minimize the generation of nitrous oxide.

Accordingly, although the manufacturing cost of vehicles using the charger technology of the present invention is

somewhat raised, it is considerably less costly to improve engine performance using the inventive charger than to simply increase engine displacement. Further, substantial value-added benefits are realized in the vehicle using the present invention.

While this invention has been described in connection with what is presently considered to be the most practical preferred embodiments, it is to be understood that the invention is not limited to the disclosed embodiments, but, on the contrary, is intended to cover various modifications and equivalent arrangements included within the spirit and scope of the appended claims.

WHAT IS CLAIMED IS:

1. A charger for an internal-combustion engine comprising:

a compressor for compressing supplied air to a high temperature/high pressure state and emitting the same;

a heat exchanger for reducing a temperature of the high temperature/high pressure air received from the compressor to convert the same into a low temperature/high pressure state; and

a turbine for expanding the air received in a low temperature/high pressure state from the heat exchanger to perform work externally such that the temperature of the air is further reduced, after which the air is supplied to the internal-combustion engine.

2. The charger according to claim 1, wherein the compressor and the turbine are mechanically connected, and a power transmitter is further provided, the power transmitter receiving drive power needed to drive the compressor.

3. The charger according to claim 1, wherein a nozzle area of the turbine is variable, and the turbine is able to perform a function of a throttle valve.

4. The charger according to claim 1, wherein the compressor and the turbine are mechanically connected, and an additional compressor is provided.

FIG. 1

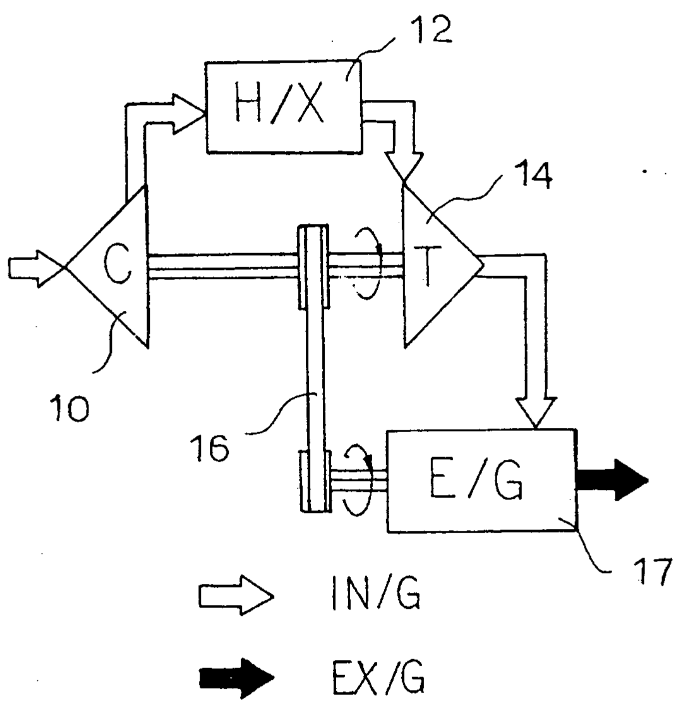


FIG. 2

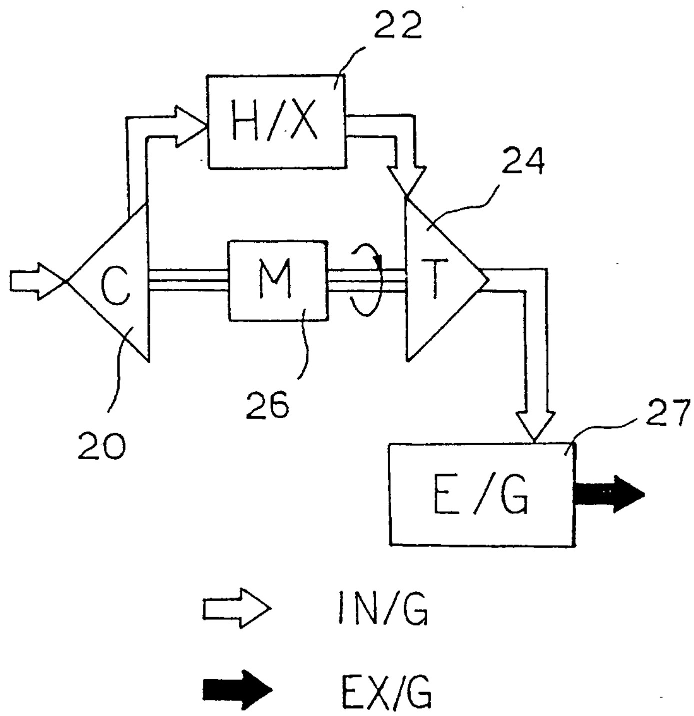


FIG.3

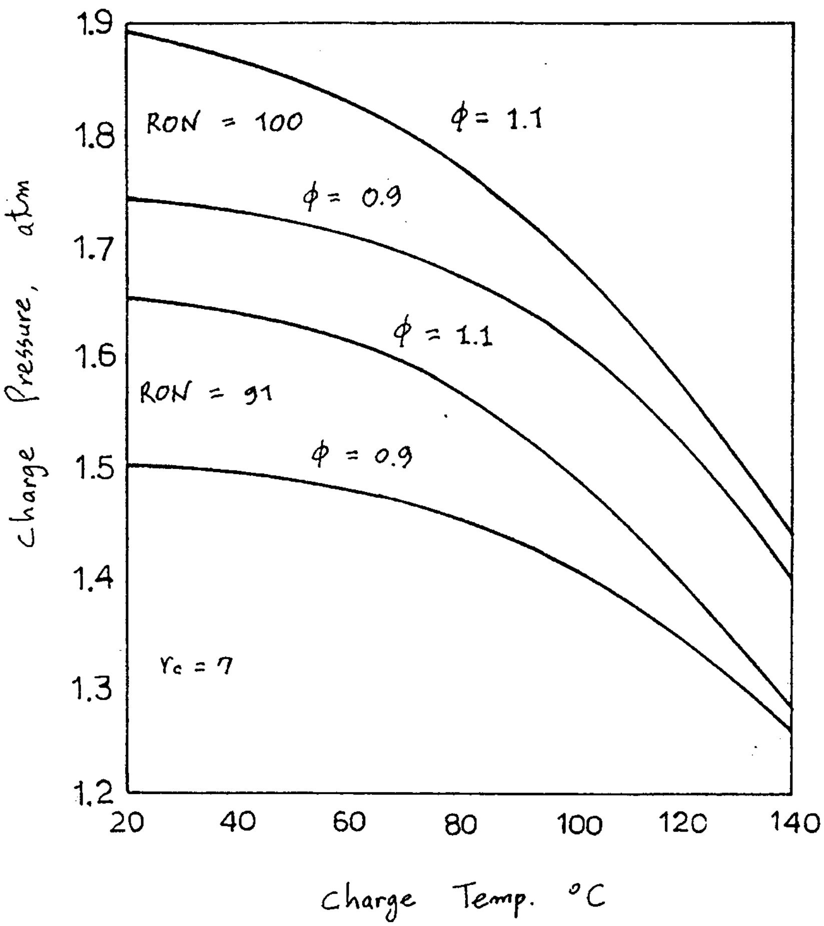

1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
20
40
60
80
100
120
140
Charge Pressure, atm
Charge Temp. °C
RON = 100
ϕ = 1.1
ϕ = 0.9
ϕ = 1.1
RON = 91
ϕ = 0.9
r_c = 7

FIG. 4

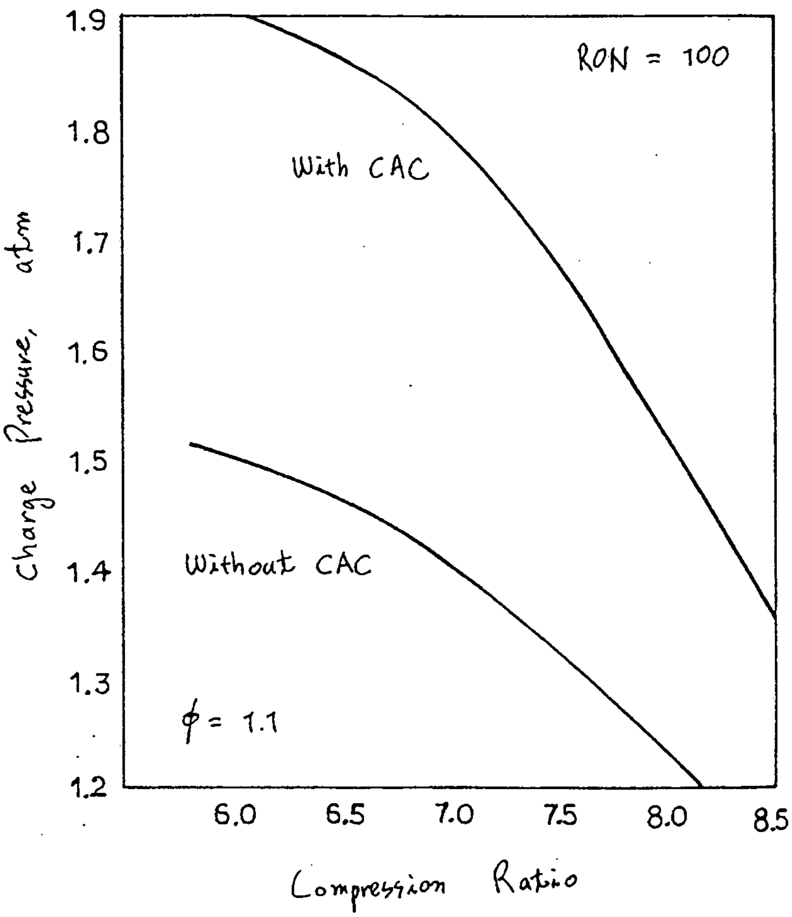
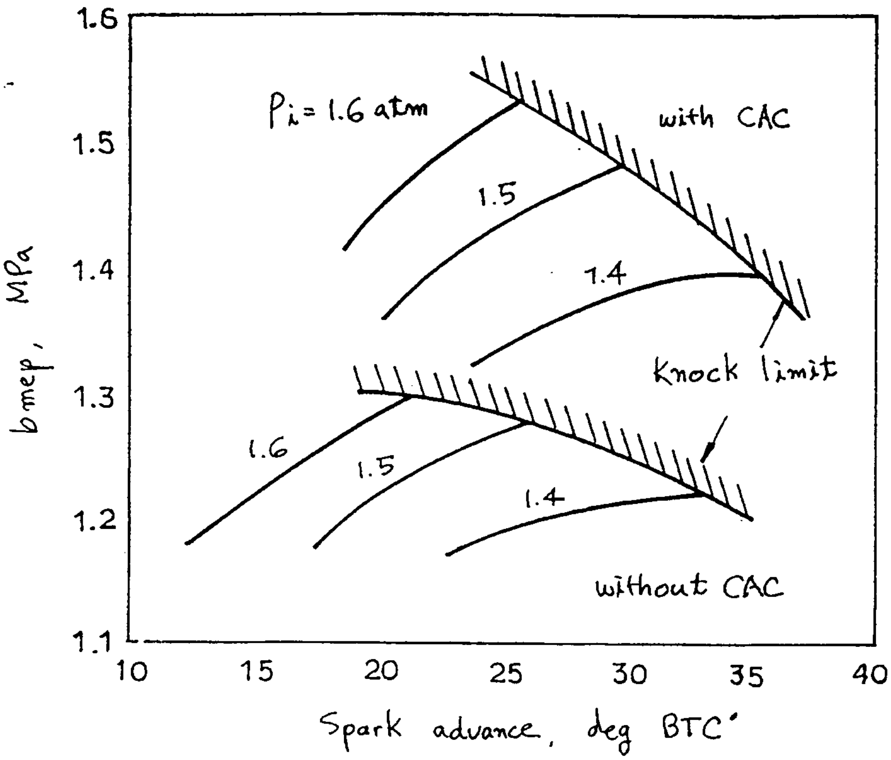

RON = 100
With CAC
Without CAC
ϕ = 1.1
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
Charge Pressure, atm
6.0
6.5
7.0
7.5
8.0
8.5
Compression Ratio

FIG. 5



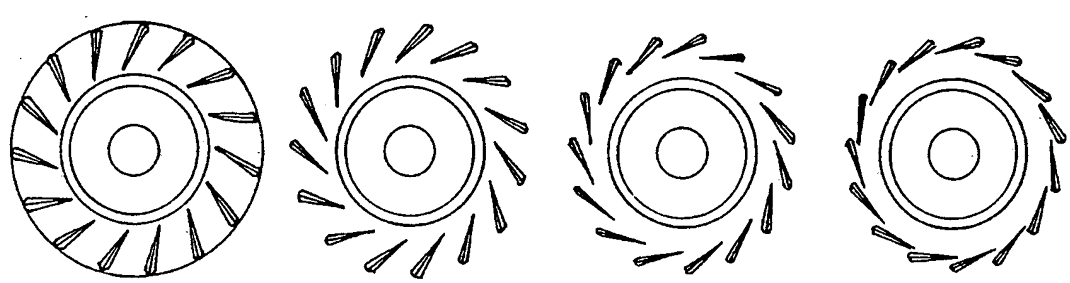


FIG.6a FIG.6b FIG.6c FIG.6d

FIG.7

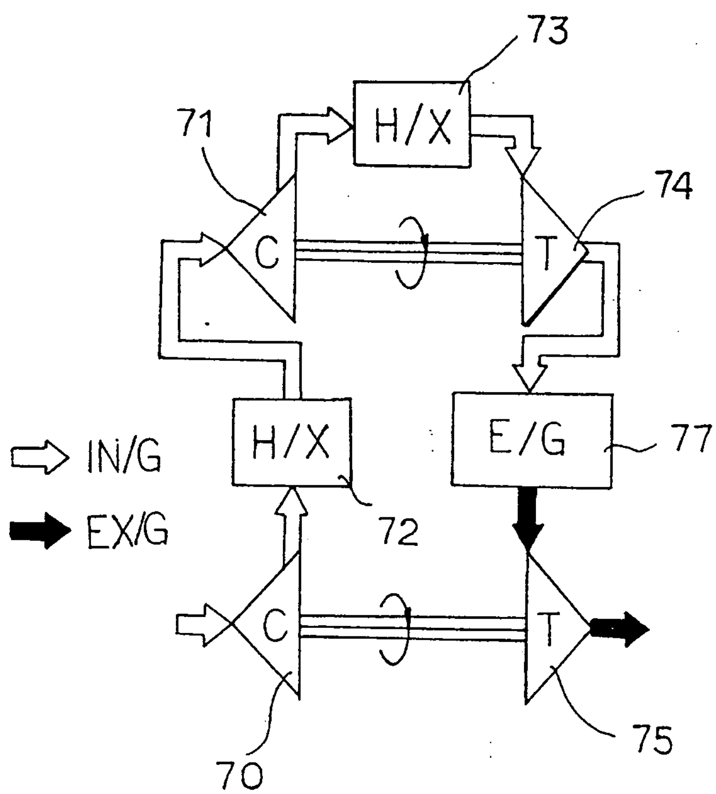
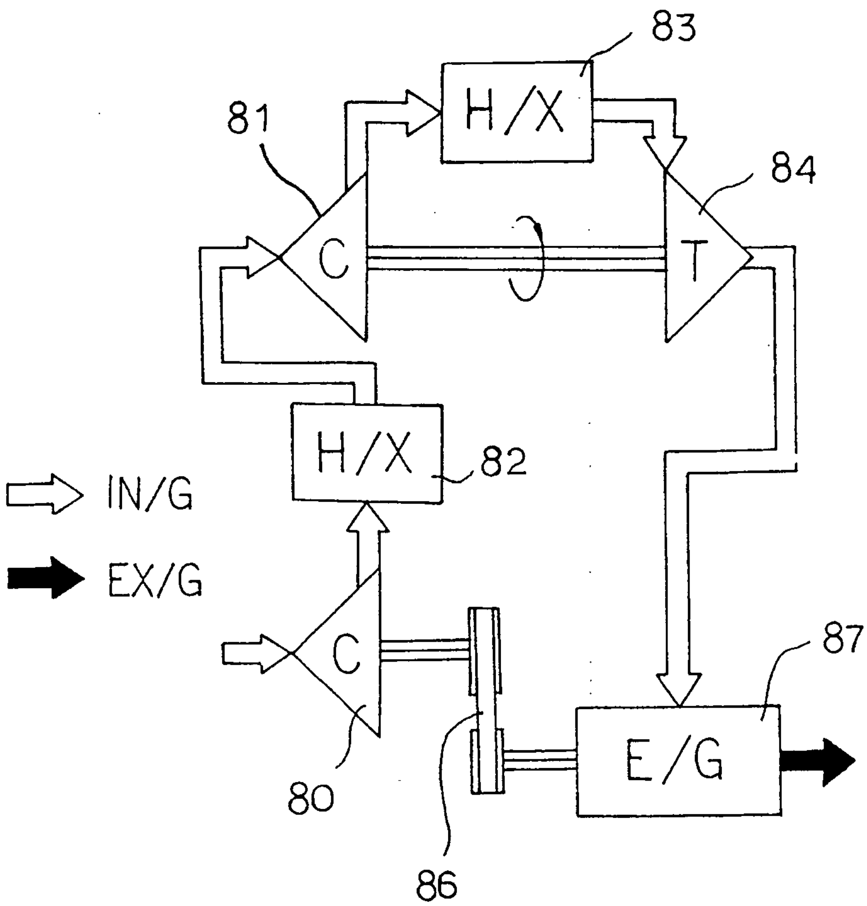


FIG. 8



INTERNATIONAL SEARCH REPORT

International application No.
PCT/KR 98/00141

A. CLASSIFICATION OF SUBJECT MATTER

IPC^6: F 02 B 29/04

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC^6: F 02 B 29/04,33/32,33/34

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPODOC, WPIL, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 2 703 561 A (FROEHLICH) 08 March 1955 (08.03.55), especially column 2, line 40 - column 3, line 3.	1-4
A	US 2 877 622 A (ANTONISSEN) 17 March 1959 (17.03.59), especially fig.1,3,4; column 2, lines 4-71.	1-4
A	FR 786 194 A (ESCHER WYSS) 28 August 1935 (28.08.35), especially claims 7,8; reference numbers C,7,13.	1-4
A	US 2 703 560 A (LIEBERHERR) 08 March 1955 (08.03.55), especially column 1, line 37 - column 2, line 47.	1-4

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
29 September 1998 (29.09.98)	15 October 1998 (15.10.98)

Name and mailing address of the ISA/	Authorized officer
Austrian Patent Office Kohlmarkt 8-10; A-1014 Vienna Facsimile No. 1/53424/535	Fietz Telephone No. 1/53424/358

INTERNATIONAL SEARCH REPORT

Inform. .n on patent family members

Inte .onal application No.
PCT/KR 98/00141

Im Recherchenbericht angeführtes Patentdokument Patent document cited in search report Document de brevet cité dans le rapport de recherche	Datum der Veröffentlichung Publication date Date de publication	Mitglied(er) der Patentfamilie Patent family member(s) Membre(s) de la famille de brevets	Datum der Veröffentlichung Publication date Date de publication
US A 2703561		keine - none - rien	
US A 2877622		keine - none - rien	
FR A 786194		keine - none - rien	
US A 2703560		keine - none - rien	